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CALCULATIONS OF INTAKE-AIR COOLING RESULTING FROM WATER INJECTION AND OF WATER RECOVERY FROM EXHAUST GAS

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SUMMARY

In connection with engine tests made by the National Advisory Committee for Aeronautics of the induction of water with the inlet air as a means of internal cooling of aircraft-engine cylinders, some preliminary calculations have been made for the effects of water in cooling the inlet air and also for the temperatures to which the exhaust gas would have to be cooled in order to recover sufficient water for this internal engine cooling. The estimates indicate that the cooling effect of the water on the inlet air can be more extensive than the cooling now obtained with intercoolers or aftercoolers in the air-induction system. In connection with water recovery from the exhaust gas, the estimates indicate that sufficient water can be recovered from 50 percent of the exhaust gas to provide an inducted water-fuel ratio of 0.5.

INTRODUCTION

In reference 1 data are presented on the use of water as an internal engine coolant to suppress knock and to maintain permissible engine temperatures at higher power outputs than are permitted with present engine cooling. After the data had been obtained from the tests reported in reference 1, estimates were made at Langley Memorial Aeronautical Laboratory of the effect of water in cooling the incoming charge and of the temperature requirements of an apparatus to recover from the exhaust gas the water required for this internal cooling. (The results of these estimates, presented herein, are not considered complete but are considered to be preliminary estimates that will provide a basis for a more extensive investigation of this problem.)

EFFECT OF INTERNAL COOLANT ON STATE OF INCOMING CHARGE

Estimates of Partial Volume of Air for Different Amounts of Water Vaporized in the Inlet Air

Figure 1 shows the relation of the fuel-air ratio F/A and the water-air ratio W/A for complete vaporization of the water to the partial pressure of the air for constant total pressure at various water-fuel ratios. In these calculations it is assumed that the molecular weight of the air is 28.8. The ratio of the partial pressure of water vapor p_w to the partial pressure of the air p_a is therefore $\frac{W}{A} \times 1.60$, in which 1.60 is the ratio of the molecular weight of air to the molecular weight of water. The equation for determining the data in figure 1 therefore becomes:

$$\frac{p_a}{p_w + p_a} = \frac{1}{1.60 \frac{W}{A} + 1} \tag{1}$$

The values $\frac{p_a}{p_w + p_a}$ represent the specific density of the air relative to dry air for a constant total pressure and a constant temperature at various water-air or water-fuel ratios.

Estimate of Relation between Temperature and Total Pressure on Water-Air Ratio at Saturation

Figure 2 presents the relation between air temperature and water-air ratio at saturation for different total pressures. In the computation of these data, the saturated-water vaporization pressures at different temperatures were taken from reference 2.

Calculations were made as follows: At an assumed temperature of 100° F, the water-vapor pressure at saturation is 1.93 inches of mercury absolute. The water-air ratio at saturation is given by

$$\frac{W}{A} = \frac{1.93 \times 18}{(p - 1.93) \times 28.8} \tag{2}$$

in which p is the total pressure in inches of mercury absolute and 18 is the molecular weight of water. If the water-air ratio given in figure 2 is assumed to represent the water vaporized for intercooling of the charge between the supercharger and the engine, it must necessarily be assumed that the inlet air was dry previous to the induction of the water.

In figure 3 additional curves are presented in which it is assumed that the air was at various degrees of humidity before the water was injected into the inlet air. In this case the water-air ratio at the specified humidity and pressure is subtracted from the water-air ratio at total vaporization.

Estimate of Water Vaporization as a Means of Cooling the Inlet Air

In the calculations of water vaporization, it is assumed that the water inducted is a liquid at a temperature of 60° F. The calculations are made by equating the gain in enthalpy of the water in changing from liquid at a temperature of 60° F to a vapor at the equilibrium temperature to the loss of enthalpy of the air. For instance, with an assumed initial air temperature of 250° F, the equation becomes:

$$\frac{W}{A} = \frac{0.241\ (250 - t)}{h_{g_t} - 28.1} \tag{3}$$

where

c_p specific heat of the air at constant pressure, Btu/(pound)(°F), 0.241

t equilibrium temperature, °F

h enthalpy of water at 60° F, Btu/pound, 28.1

h_{g_t} enthalpy of water vapor at equilibrium temperature, Btu/pound

In the solution of equation (3), different values of t are used and the values of W/A are computed. The results are shown in figure 4.

The data in figure 4 represent the degree of cooling of the inlet air for different water-air ratios, assuming complete vaporization of the water. If these curves are superimposed on the curves in figures 2 and 3, data that show the final temperature of the air at saturation for various outlet temperatures and pressures from the supercharger are determined. (See figs. 5 and 6.) In each case the intersection of the curves of temperature of the air from the supercharger and the saturation curves represents the final temperature of the mixture of air and water vapor. These intersections are plotted in figure 7, which shows that appreciable intercooling or aftercooling of the air from the supercharger is to be obtained from the vaporization of the inducted water, even for air of high humidity. The data indicate that, through the use of water injection, intercooling or aftercooling in the supercharger might be eliminated. Consequently, the additional weight of a water-recovery apparatus for all or part of the exhaust gases might be offset by the elimination of the supercharger intercooler or aftercooler.

Computations made from the data in figures 1 and 7 and the results presented in figures 8 and 9 show the increase in density of the incoming air that results from cooling the air by vaporization of the water to the saturation point. In the engine tests presented in reference 1, water-fuel ratios used were greater than the ratios reached at saturation. In these cases the vaporization of the water injected into the induction system would also be dependent upon the point and manner of injection, and the completion of evaporation of this water would take place within the cylinder thus internally cooling the engine by absorbing heat during the compression stroke or during the combustion process. Consequently the total effective cooling that can be obtained by injecting water into the incoming charge may be greater than that given in the preceding figures.

The heat required to vaporize the portion of the water vaporized before the mixture is inducted into the cylinder does not enter into the efficiency of the engine. The heat of vaporization extracted from the air during the compression stroke must be considered in the estimate of the efficiency of the engine.

TEMPERATURE REQUIREMENTS OF WATER-RECOVERY APPARATUS

Estimate of Partial Pressure of Water in Exhaust Gas

According to reference 3, at a fuel-air ratio of 0.067, the water content of the exhaust gas is 14.1 percent by volume of the dry exhaust gas. This value is equivalent to 12.3 percent of the total pressure of the exhaust gases and the ratio of the water formed is 1.25 times the weight of fuel inducted. If water is introduced into the inlet manifold, the percentage of total water in the exhaust gas will, of course, increase. Estimates of partial pressures of water in the exhaust gas for different quantities of water inducted with the inlet charge are presented in the following table:

Water/fuel in exhaust	Water/fuel in inlet	Water in dry exhaust (percent by volume)	Partial pressure of water in exhaust (percent)
1.25	0.0	14.1	12.3
1.75	.5	19.7	16.5
2.25	1.0	25.4	20.3
2.75	1.5	31.0	23.7
3.25	2.0	36.7	26.8

The curves for the partial pressure of the total water content under these conditions are shown in figure 10. The line W/F = 0 represents the partial pressure of the water formed in the exhaust gases.

Saturation Temperatures of Exhaust Gas

The first problem in the recovery of water for recirculation in the engine is the determination of the temperature to which the exhaust gas will have to be cooled in order to recondense the water. If water is recirculated from the exhaust gas to the inlet manifold, only the portion of the water that is to be used for the recirculation need be recovered; that is, after the system has once reached equilbrium, the water formed by combustion need not be recovered except to replace losses and possibly to provide for elimination of condensed moisture trails.

Figure 11 presents the relationship between the temperature and the saturated vapor pressure of water. From the curve for a water-fuel ratio of 0 in figure 10, partial pressures at the different total pressures are chosen and the corresponding saturation temperatures are determined from figure 11. The saturation temperatures at each pressure are taken from reference 2. From a standard altitude table (reference 4) the altitudes giving atmospheric pressure equivalent to the total pressures are also determined. If these altitudes are plotted against the corresponding saturation temperatures (fig. 12), the percentage of water of combustion formed at each altitude is determined. The curve for 100 percent of the water of combustion causing saturation represents the temperature at each altitude to which the exhaust gas will have to be cooled to condense all water except the water formed during the combustion process, provided that the pressure of the exhaust gas is the pressure of the ambient air.

This 100-percent curve in figure 12 does not allow for any water to be lost; that is, the assumption is made that all the water recovered is inducted into the inlet air and an equilibrium is established in regard to the water being circulated. An estimate of the required temperatures to which the exhaust gas will have to be cooled to provide additional water condensation to cover losses is made from the two curves in figure 12 for 50 percent and 25 percent of the water of combustion causing saturation. These curves indicate at each altitude the temperature to which the exhaust gas will have to be cooled in order to condense, in addition to the water that is being recirculated to the inlet air, 50 percent and 75 percent of the water formed by the combustion process.

The curves of figure 12 represent the maximum allowable temperature to which the exhaust gas must be cooled. The data show that at high altitudes there is not too much difference between the freezing temperature and the temperature required for condensation of the water.

Estimate of Effect of Percentage of Exhaust Gas Passed through Water Recovery Equipment on Total Water Recovered

The data in figure 12 show that the temperature variation required for the condensation of different percentages of the water of combustion is small. For this reason it may be advisable, in reclaiming varying amounts of the water content of the exhaust gas, to bypass a fraction of the exhaust gas through the water recovery equipment and to extract as much water as possible from this fraction.

If it is assumed that from the gas passed through the water recovery equipment, a mass of water is recovered equal to the mass of all the injection water contained in that gas plus a certain fraction of the water of combustion contained therein, then

$$W_T = m\ (W_I + nW_C)$$

where

W_T total water recovered

W_I water inducted into inlet manifold

W_C water formed by combustion

m percentage of exhaust gas passed through aftercooler

n percentage of water of combustion recovered

At equilibrium $W_I = W_T$. Therefore

$$\frac{W_T}{W_C} = \frac{-mn}{1 - m}$$

If the values of the ratio W_T/W_C are multiplied by 1.25, the ratio of water of combustion to fuel as given in reference 3 at a fuel-air ratio of 0.067, the equilibrium values for the ratio of water recovered to fuel inducted are given. These values are plotted in figure 13. If 50 percent of the exhaust gas is passed through the water recovery equipment and the water is recovered down to 50 percent of the water formed by combustion, the water-fuel ratio is 0.6, a value that is sufficiently below a knock limit to reduce the octane requirement of an engine burning 100-octane fuel to 80-octane fuel (reference 1).

In connection with the use of turbosuperchargers, it is pointed out that the data presented in figure 7 of reference 1 indicate that introducing water into the incoming charge did not lower the exhaust-gas temperature although admittedly the facilities for the measurement of exhaust-gas temperatures were not satisfactory.

Based on these data the use of water induction as an internal coolant will probably not result in difficulties when the power plant also employs a turbosupercharger. Actually a temperature drop in the exhaust gas prior to its passage through the turbosupercharger is permissible.

CONCLUSIONS

The estimates presented herein indicate:

1. The use of water as an internal coolant provides a method for obtaining an appreciable temperature drop in the incoming charge.

2. The use of water as an internal coolant appreciably increases the quantity of air inducted into the engine if vaporization to saturation is completed before the charge enters the engine.

3. Water-fuel ratios up to 0.5 can be achieved by the recovery of water from one half the exhaust gas.

4. It is apparent that certain practical problems will arise in connection with the application of water recovery equipment, but the study of these phases is beyond the scope of the present paper.

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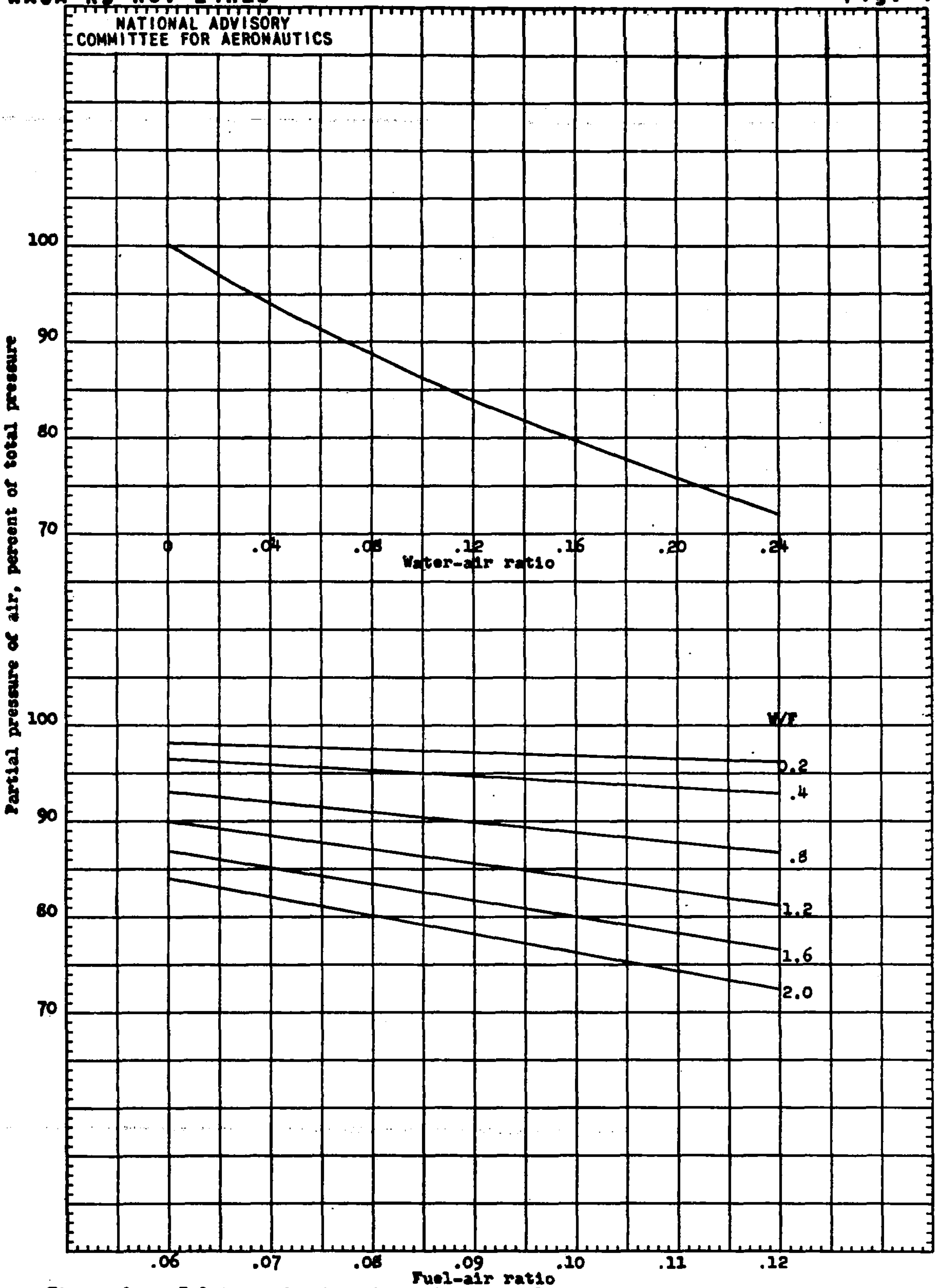

Figure 1. - Relation of water-air ratio and fuel-air ratio for complete vaporization of water to partial pressure of air for constant total pressure.

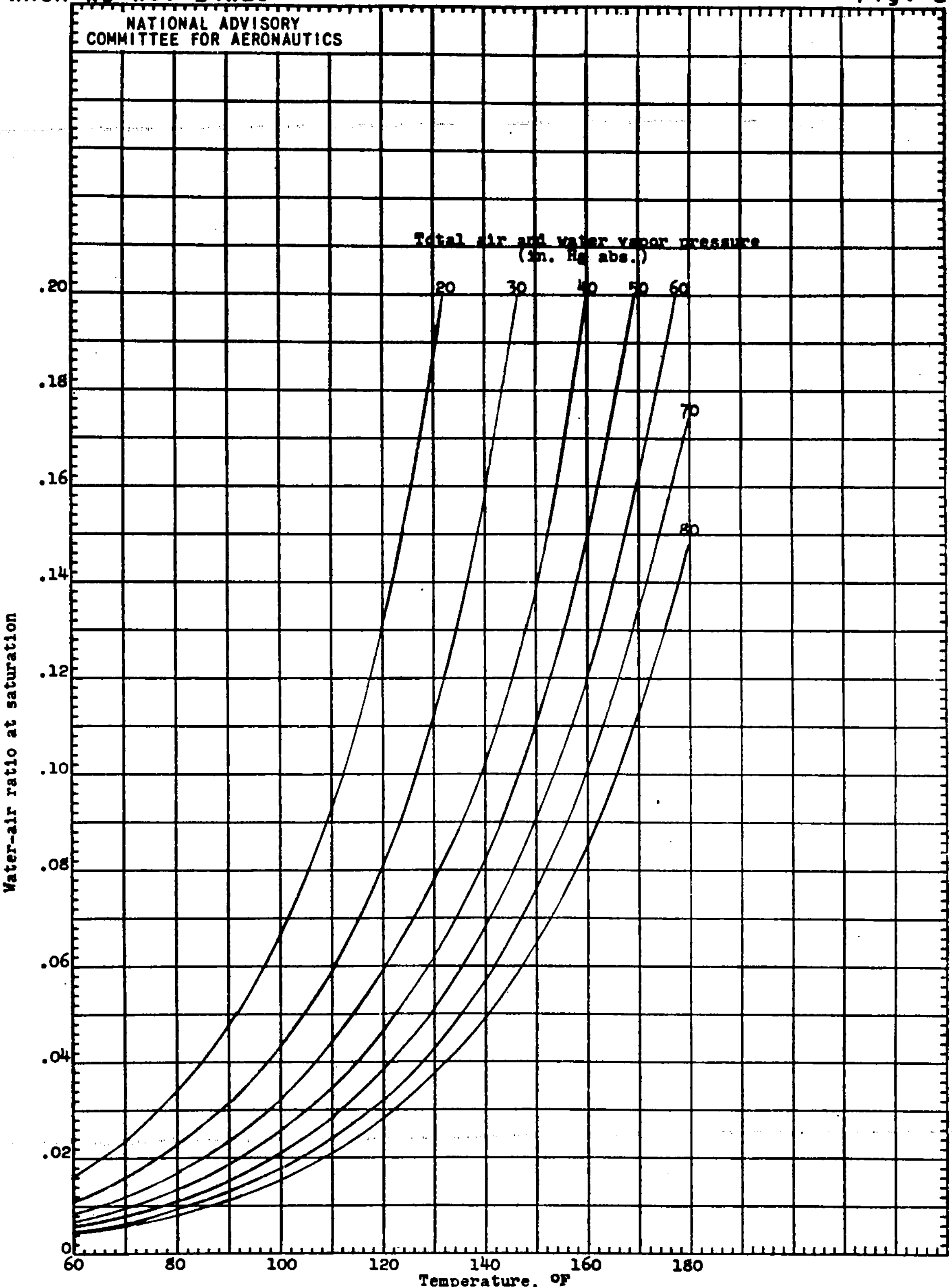

Figure 2. - Relation between temperature, total pressure, and water-air ratio at saturation. Air dried before induction of water. Water inducted as a liquid at 60° F.

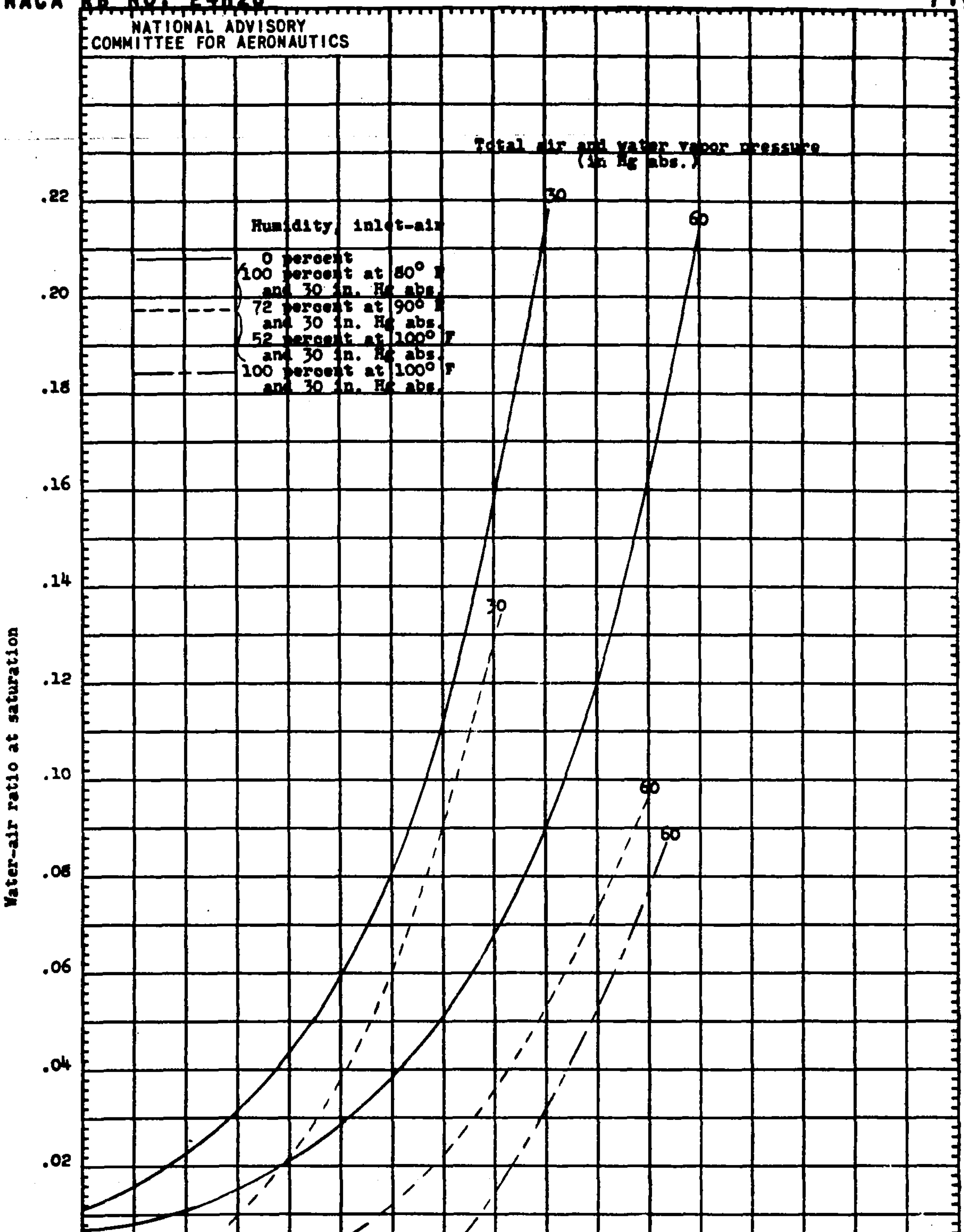

Figure 3. - Relation between temperature, total pressure, and water-air ratio at saturation for different values of humidity before water is injected into air. The water-air ratio represents the water vaporized to cause saturation in addition to the water already contained in the air. Water inducted as a liquid at 60° F.

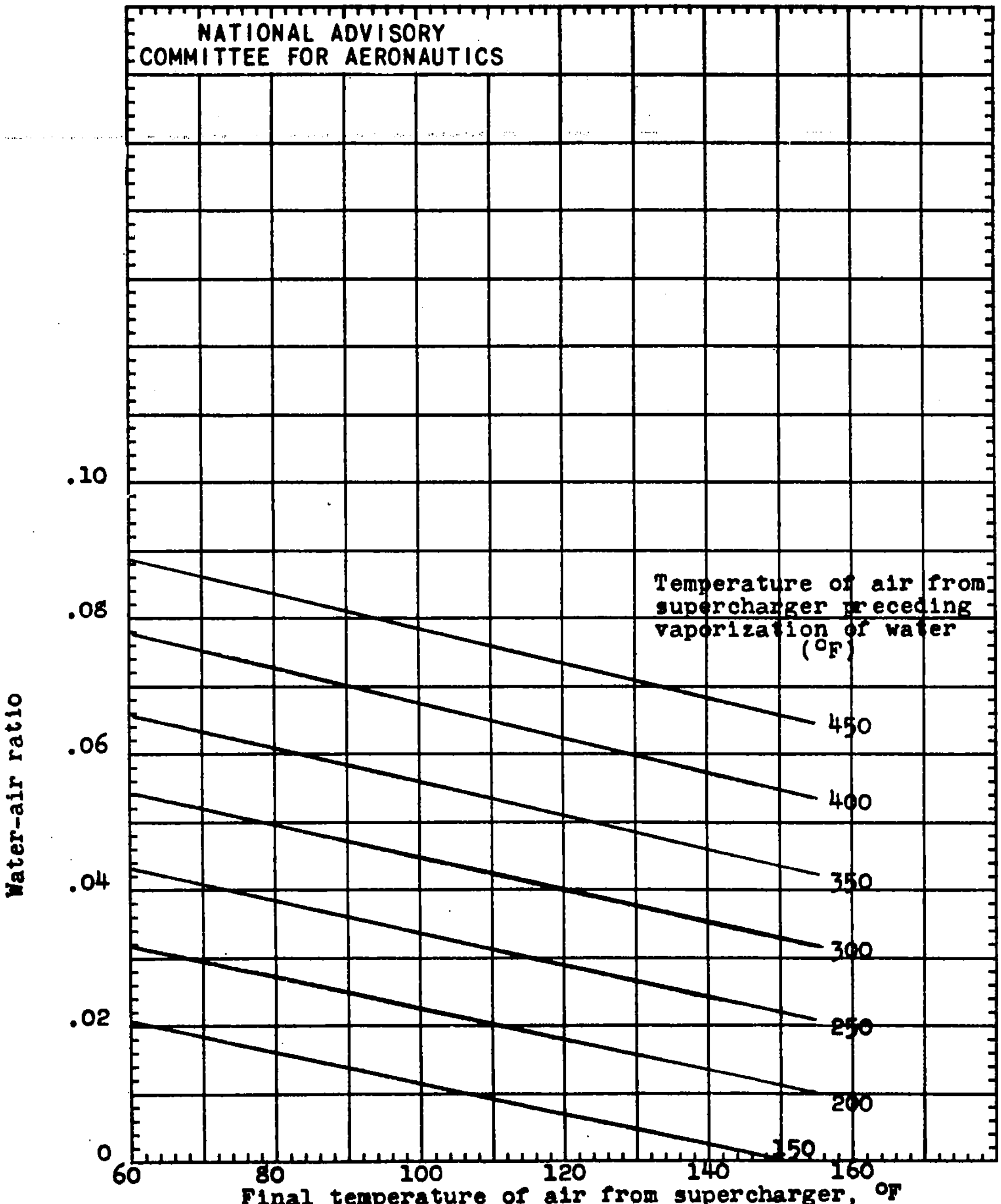

Figure 4 . - Relation between final temperature of air from supercharger after vaporization of water and amount of water vaporized, expressed as water-air ratio, for different outlet temperatures from the supercharger. Water inducted as a liquid at 60° F.

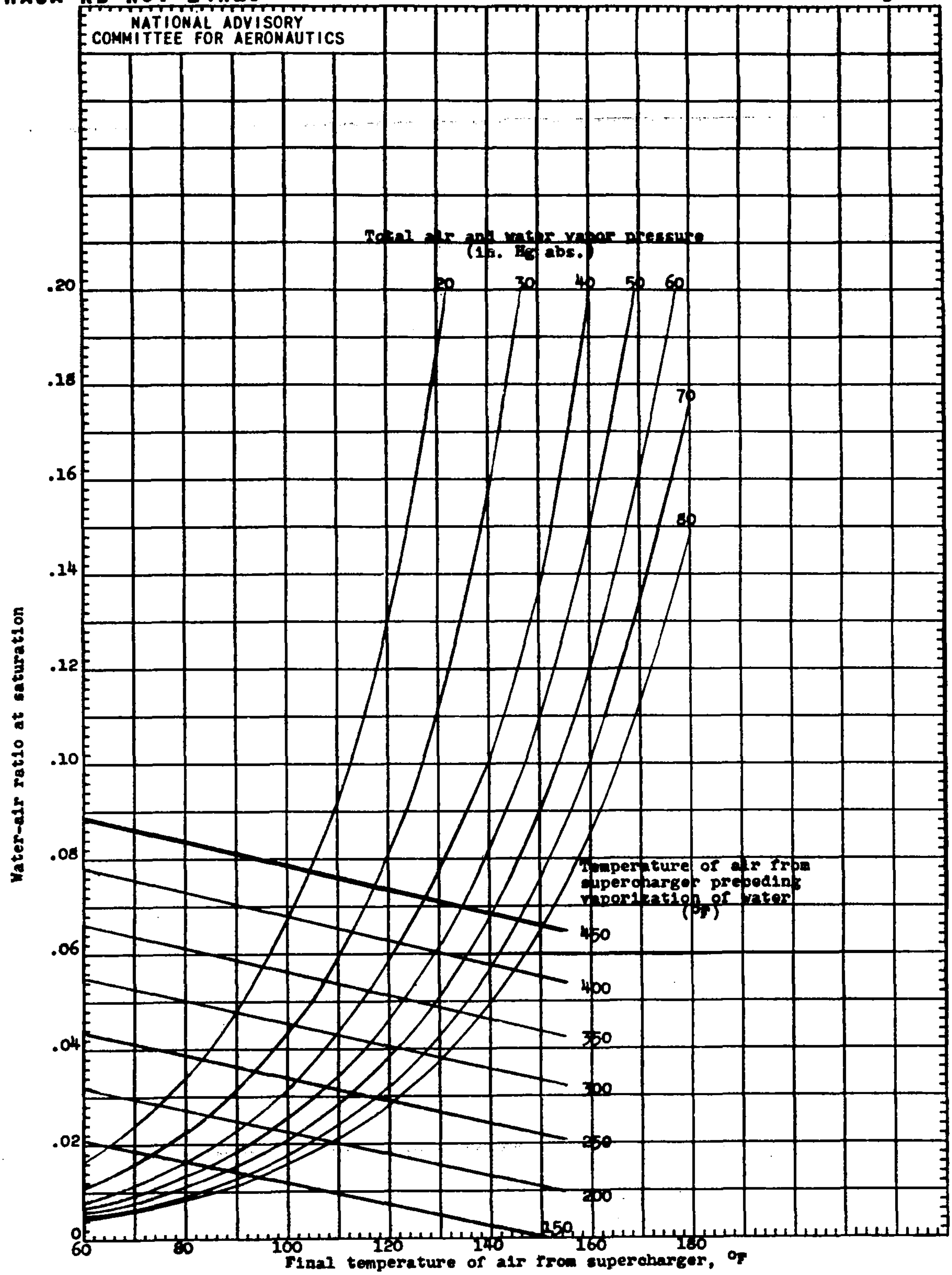

Figure 5. - Relationship of final temperatures of air from supercharger cooled by vaporization of different quantities of water at saturation. Air dried before introduction of water. Water inducted as a liquid at 60° F.

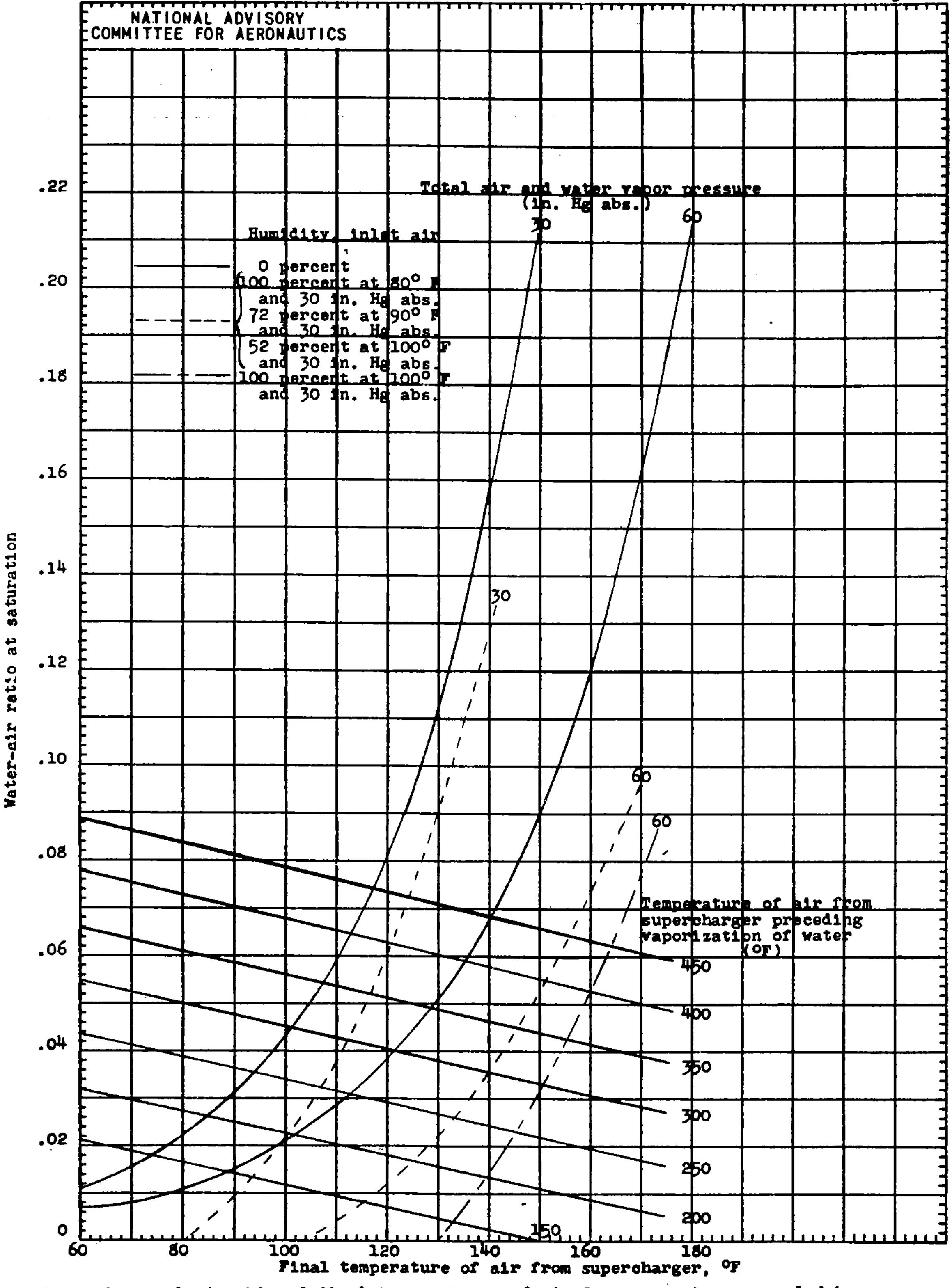

Figure 6. - Relationship of final temperatures of air from supercharger cooled by vaporization of different quantities of water at saturation. Different humidities for air before induction of cooling water are assumed. Water inducted as a liquid at 60° F.

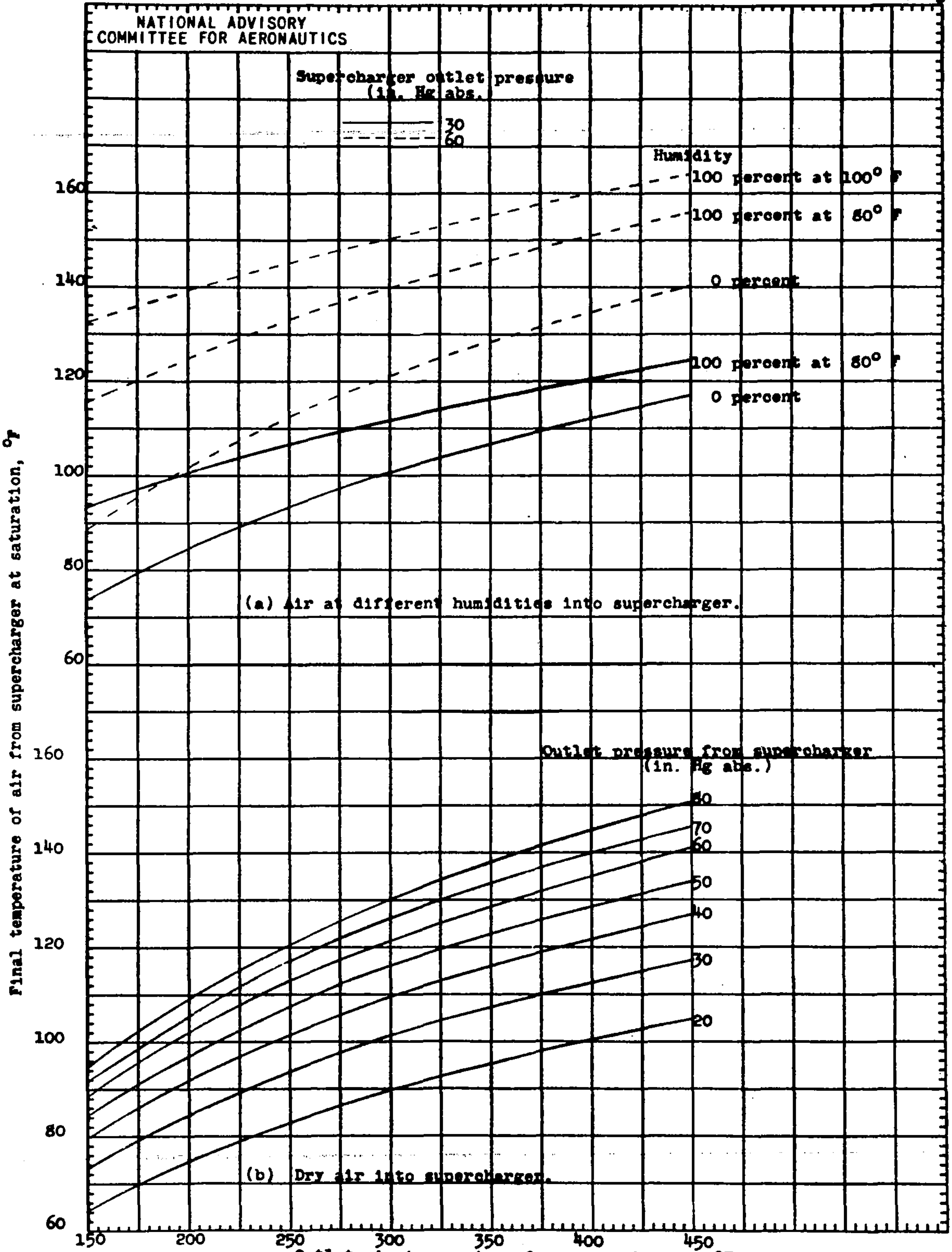

Figure 7. - Relation between outlet-air temperature from supercharger and final temperature of air when sufficient cooling water is inducted for saturation of the air. Water inducted as a liquid at 60° F.

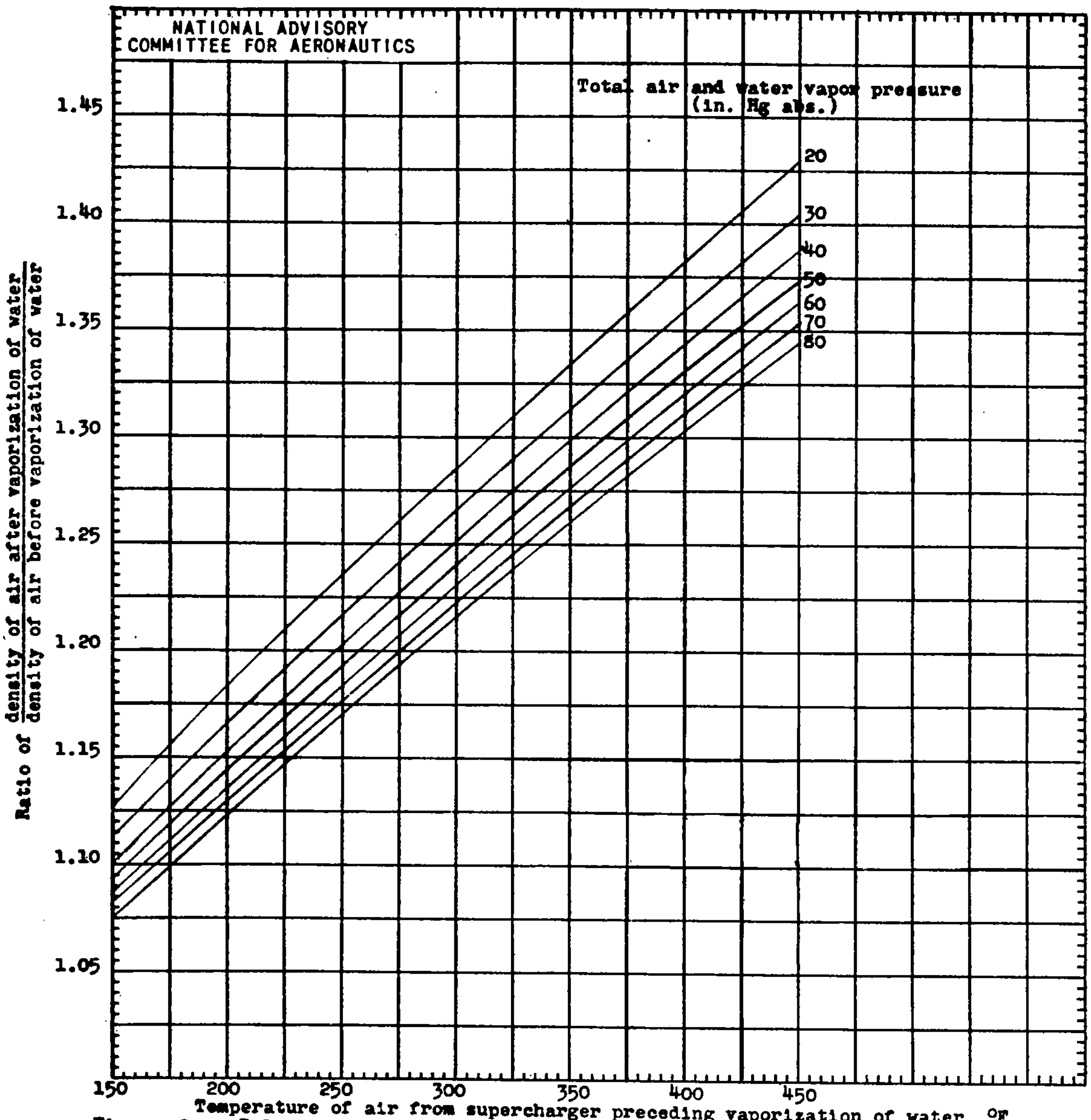

Figure 8. - Relation between inlet-air temperature preceding introduction of water and density increase of air at the final air-water mixture temperature for saturation.

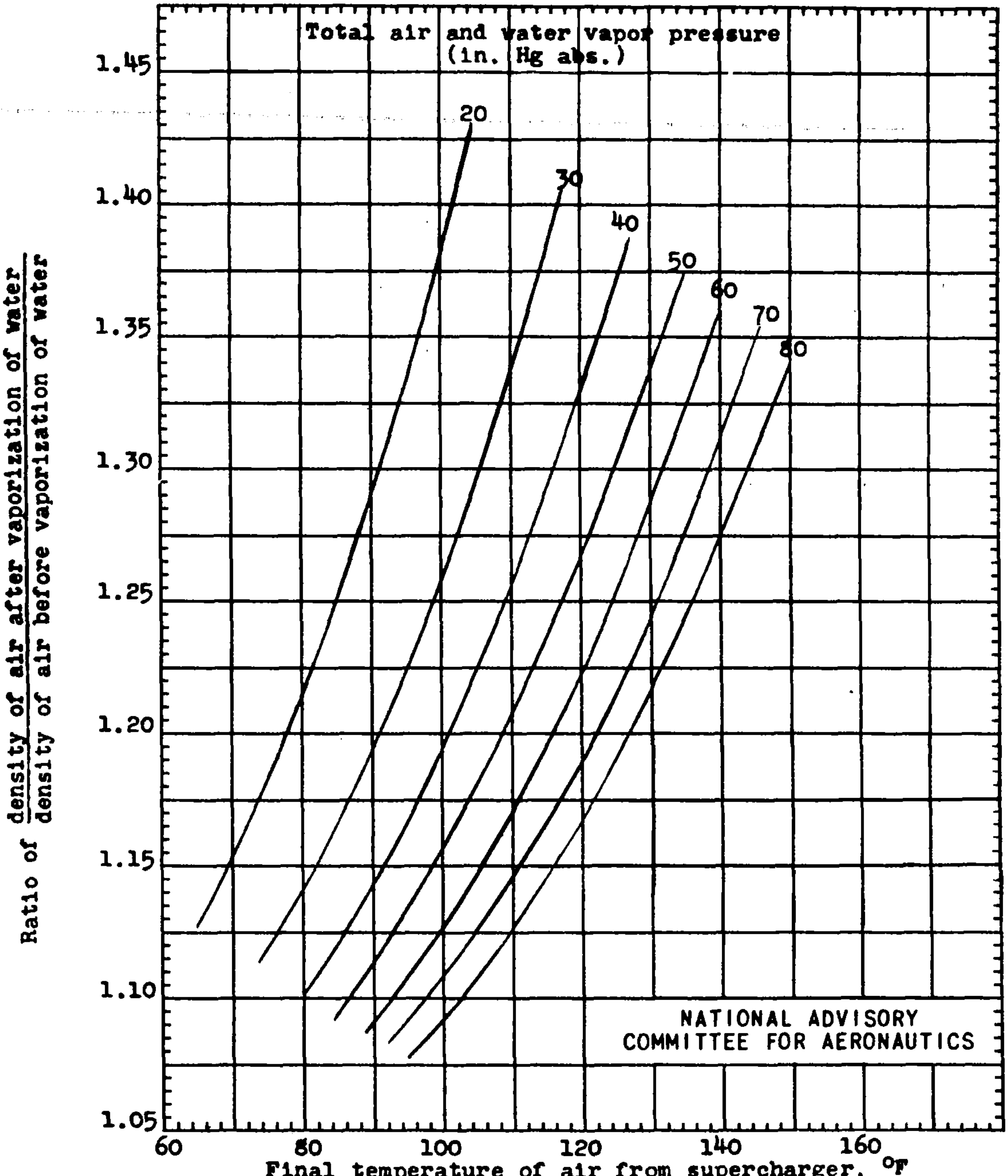

Figure 9. - Relation between air-water saturation temperature and increase of air density through vaporization of water to saturation point.

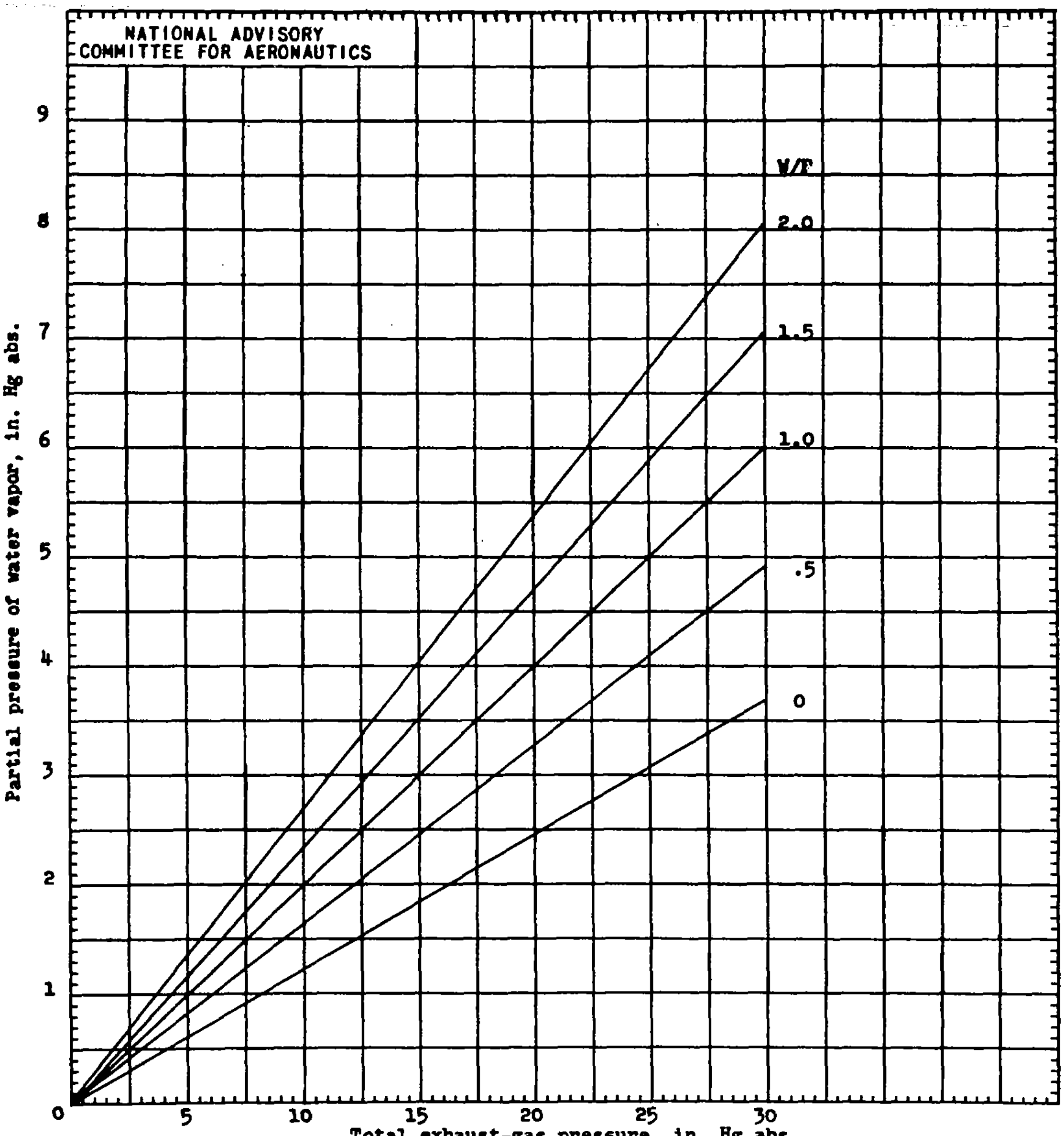

Figure 10. - Relationship between total pressure of exhaust gas and partial pressure of water vapor in the exhaust gas for different quantities of water inducted with the inlet-air. The water-fuel ratio W/F represents the ratio of inducted water to inducted fuel. Fuel-air ratio, 0.067.

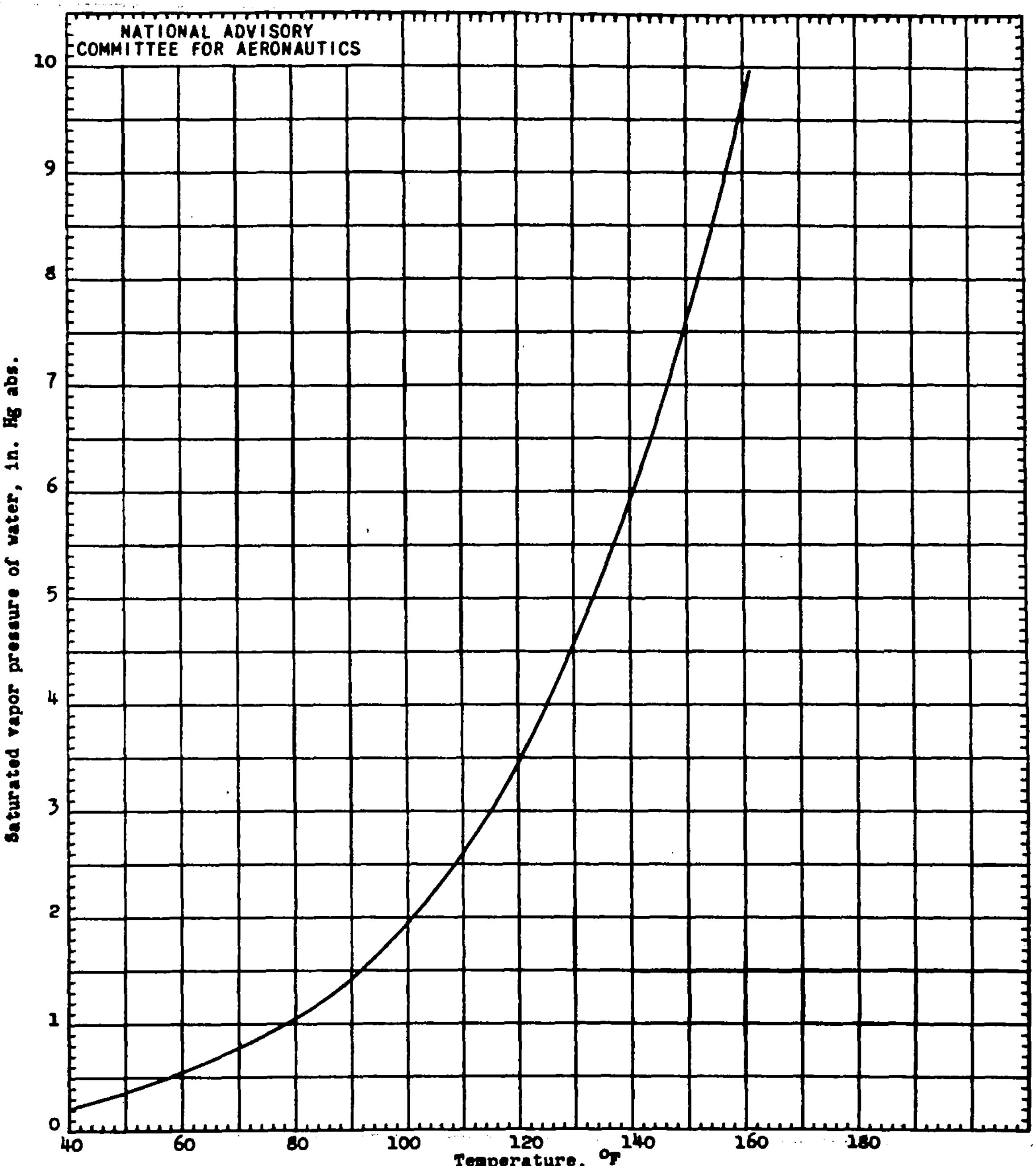

Figure 11. - Relation between temperature and saturated vapor pressure of water.

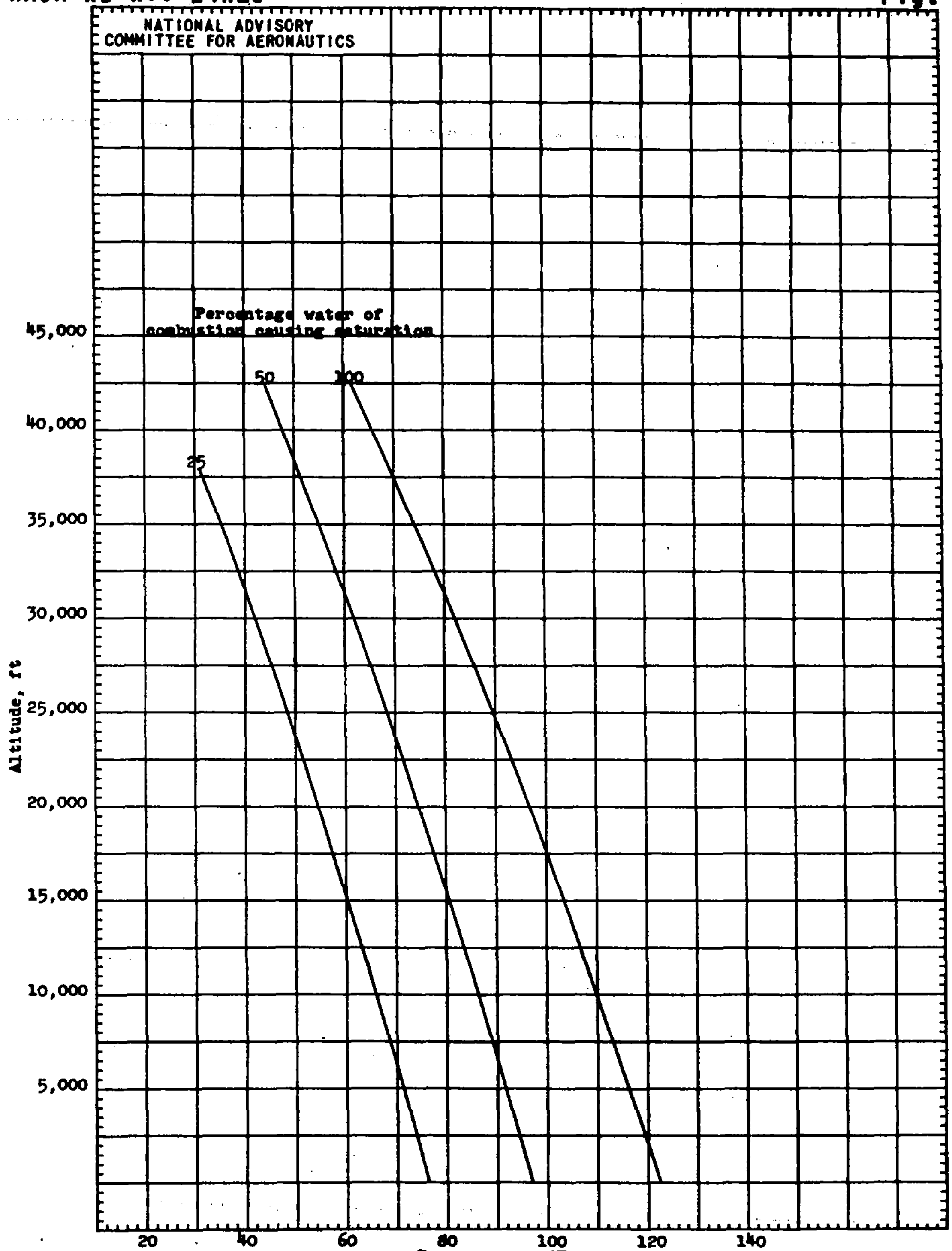

Figure 12. - Relation between exhaust-gas saturates and altitude. Fuel-air ratio, 0.067. It is assumed that the exhaust gas is at the ambient pressure at the indicated altitude.